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ABSTRACT

The paper describes a new approach used to simulate the vibration in high frequency (up to 4 kHz) of an hermetic compressor for domestic appliances.
The described methodology has been applied to evaluate noise emission and optimize the compressor design at the first stage of its development.
The approach can be applied to evaluate noise emission due to structural forces, considering contacts and impacts of mechanical components, and has been successfully applied to redesign a compressor with optimized noise emission.

1. INTRODUCTION

During the development of a new product, which showed an unpredicted problem in the 3150 Hz third octave band, an intense campaign of measurements was performed, in order to identify the noise source.
This source was particular, as it showed an unique peak at an unique frequency (2950 Hz).
A procedure developed in order to eliminate this high-frequency problem through the compressors' components' re-design is described in this paper.
It allowed us to determine which component (or which coupling between components) was critical to noise and to study the problem's dependence from the components' design, thus giving the chance to determine the best low-noise-related components' assembly.

2. EXPERIMENTAL MEASUREMENTS

2.1 Acceleration Measurements

A set of acceleration measurements was performed, in order to identify the noise source through the evaluation of possible anomalous values for some parameters.
The accelerations were measured in different points on the compressor pump (Fig.1), including some points on the crankcase, on the cylinder head, on the discharge tube and on the brackets.
The signals were acquired and then processed with Lms Cada-X software, in order to get their frequency content.
A peak value at the same frequency of the overall compressor's emitted noise was found only in two positions, corresponding to:

1. the cylinder head "front" (point 4 in Fig.1)
2. the cylinder head "rear" (point 2 in Fig. 1)

These points were then chosen as "performance indicators" for the simulations later performed: whenever the simulated value was lower than the previously-obtained ones, the new configuration was assumed better than the previous ones.

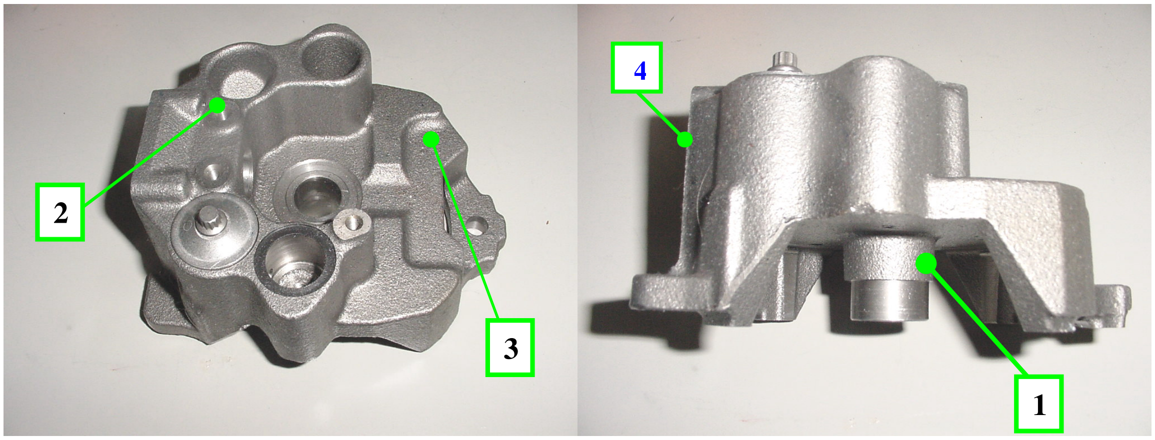

Figure 1: Some positions on the crankcase for accelerations measurement

3. NUMERICAL MODEL

A complete FE model of the pump group is assembled, starting from the single components.
This model, forced with a unitary pressure on the piston's top, is used to calculate the acceleration in the previously-described characteristic points on the crankcase and the cylinder head.

3.1 Compressor's components modal analysis and numerical validation

All the compressor's components (crankcase, connecting rod, etc.) were first numerically analyzed through modal analysis (Fig. 2) and then compared with the results (mode shapes and eigenfrequencies) obtained from experimental modal analyses.
A good correlation either was found or obtained through model updating. Also all the material properties were precisely determined and tuned, in order to get a good correlation between simulations and reality.

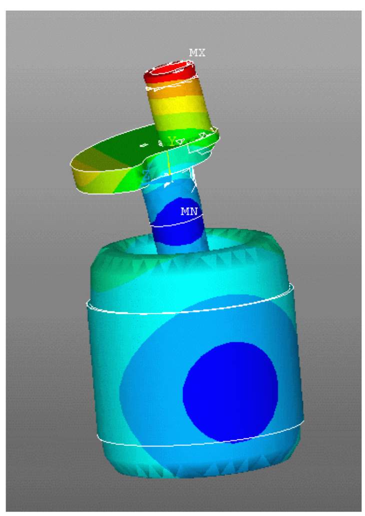

Figure 2: Rotor numerical mode

3.2 Compressor's assembly modal analysis and numerical validation

The whole compressor pump was then assembled in an unique F.E. model, a first numerical analysis was run and compared with the results obtained from an experimental modal analysis.

Although the results were already quite satisfying, some modifications in terms of material and contacts' properties were done, in order to have an even better correlation (Tab.1).

Experimental Freq [Hz]	Numerical Freq. [Hz]
543	681
697	705
1590	1579
1705	1720
2088	1945
2330	2388
2906	2881
3068	3059
3459	3422

Tab.1 : Comparison between numerical and experimental modes for the pump group

In particular the mode at 2906 Hz (experimental, 2881 Hz from the numerical analysis) was considered critical, as (Fig.3) it involved also the cylinder head zone, in an axial movement whose amplitude was much higher than the current products' one.

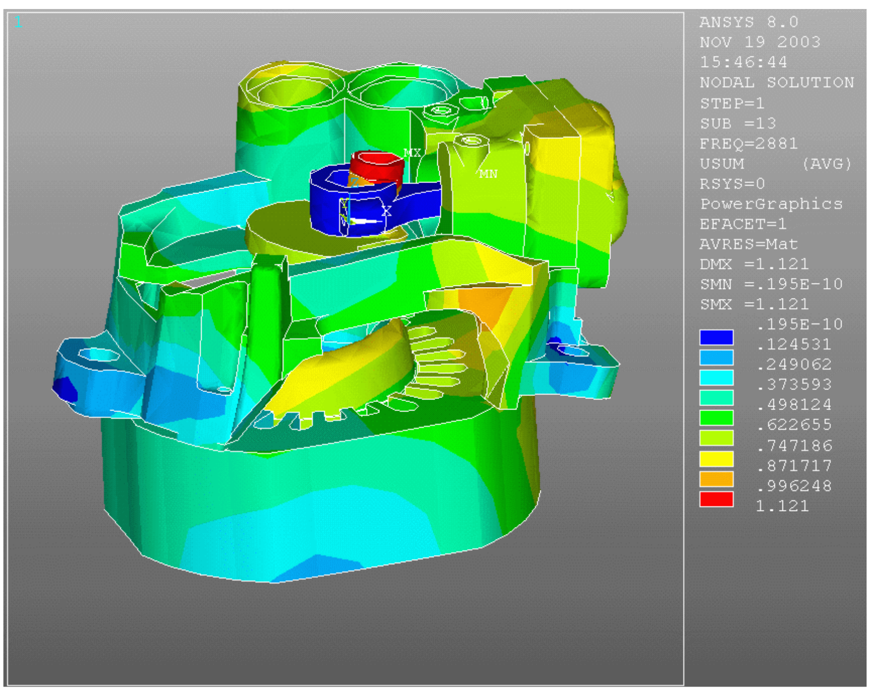

Fig. 3 : Numerical 2881 Hz mode's modeshape

This procedure was repeated for the standard model, then the material properties and the contact type were used for all the other prototypes (at that moment not built) in which the difference consisted mainly in change of geometry.

3.4 Compressor's assembly forced response analysis

The over-described model was then used for a forced harmonic analysis: on the piston's top an unitary pressure was applied and then the response in terms of frequency content was evaluated in the points which were considered significant from the first acceleration measurements.

Other F.E. models were then created changing geometrical parameters of the assembly, such as crankshaft diameter or material, connecting rod's geometry, etc.
The acceleration values obtained were then compared with the one of the original (standard) model, allowing the best choice in terms of minimizing acceleration and thus noise.

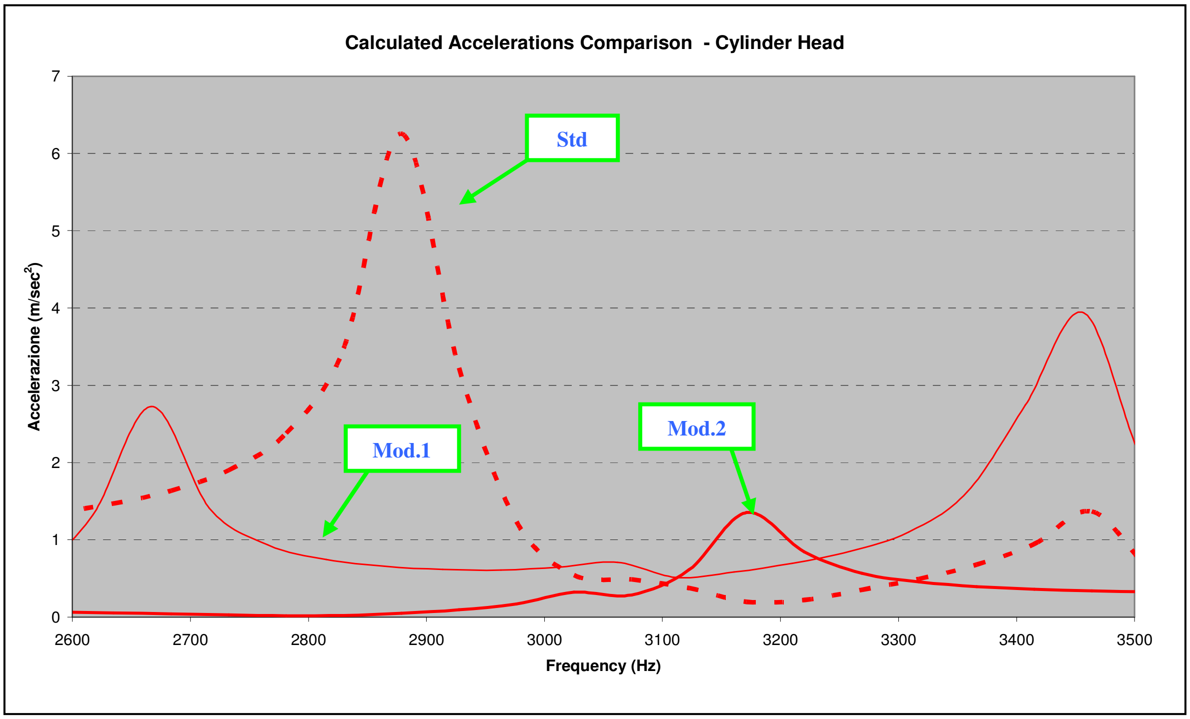

Fig.3 : Comparison between calculated acceleration between std. model and modified ones

As you can guess from Fig.3 the "Mod.2" version was chosen and then implemented.
Numerical and experimental accelerations were compared in both versions (standard and Mod.2), despite absolute value of the peak were not equal, the acceleration ratio between two versions were the same.

4. CONCLUSIONS

A F.E. model-based procedure was developed in order to solve a noise problem at 3150 Hz.
This procedure started with the assembly, calculation and validation of a F.E. model of a standard (physically built) standard compressor. The acceleration in some points of the compressors were determined and then compared with the ones obtained from other models, obtained with little geometrical modifications of the standard one.
The model with the lowest acceleration values was then chosen to be implemented.

5. REFERENCES

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